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Vibration reduction of machine tool using viscoelastic damper support

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Abstract

The damping characteristics of machine tool support system has a great influence on the rocking vibration amplitude. In our previous study, a system for machine tool support has been proposed to increase damping without decreasing stiffness. In this study, proposed damper system is applied to various machine tools. Results indicate that the damper is effective for machines larger than the machine that is already applied. However, the damping coefficient can be different between applied machines even with the same damper contact area and preload. This suggests a possible confounding factor that should be considered when determining the damping coefficient.

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Prof. Matthias Putz

Keywords: Machine tool; Damping; Vibration; Viscoelastic damper; Machine support;

1. Introduction

Residual vibration has a great influence on the tool-workpiece vibration in the machine tools [1]. Rocking vibration is the main source of the residual vibration in low frequencies. In the rocking vibration, the whole machine body moves over the machine support system [2]. Vibration characteristics between a machine bed and a floor has a great influence on rocking vibration [3]. Thus, the damping of the support system (support damping) is important to reduce the amplitude of rocking vibration.

Some damper systems have been proposed to increase the support damping [4, 5]. Most of them are active dampers. Passive dampers are more cost effective than the active dampers [1]. Although it is difficult to increase the damping without decreasing the stiffness of the support system (support stiffness) in passive dampers [3, 6, 7]. To solve this problem, a new application method of a viscoelastic damper for machine tool support has been proposed in our previous research [8]. This system is designed to increase support damping without decreasing support stiffness. This system will be detailed in the section 2.

The proposed system was effective for a small light milling machine. In this paper, the proposed system is applied to larger and heavier machines to make the system more practical. The effect of machine size to the increased damping ratio is also investigated.

2. Fundamental idea of the proposed damper

By design, damper support stiffness is lower than machine support stiffness [6, 7]. Figure 1 shows a basic damper support design in a one degree of freedom vibration system. In current support systems, the damper support is generally connected to the original machine support (stiffness support), in series as shown in Fig 1 (a). Overall support stiffness is decreased because of the lower-stiffness damper support. Increasing the stiffness of the damper support to increase overall support stiffness would compromise its damping ability [9]. In the proposed system depicted in Fig. 1,

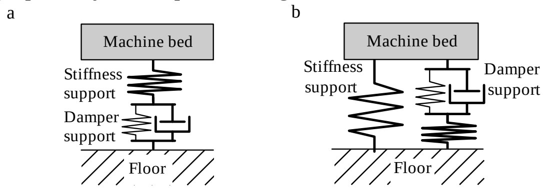

Fig. 1. Basic idea of damper; (a) Existing damper; (b) Proposed damper.

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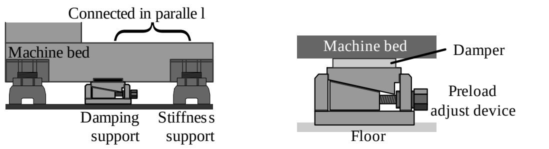

Fig. 2. Proposed damper system; (a) Overall system; (b) Damper support.

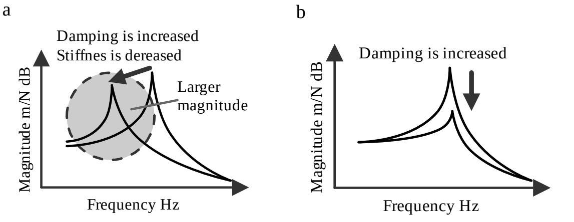

Fig. 3. Basic characteristics of damper support; (a) Existing damper; (b) Proposed damper.

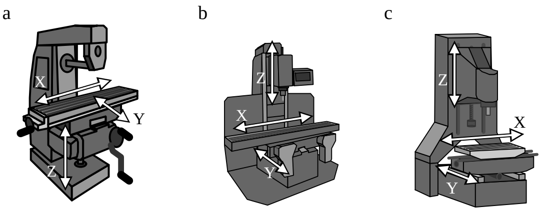

Fig. 4. Machine tools used in the experiment; (a) Small machine; (b) Medium machine; (c) Large machine.

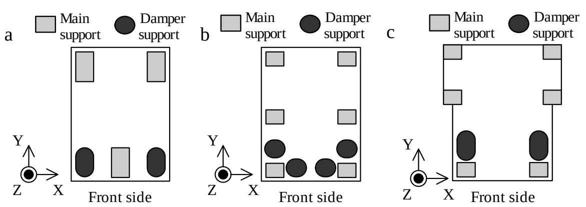

Fig. 5. Support configuration used in the experiment; (a) Small machine; (b) Medium machine; (c) Large machine.

the damper support is connected in parallel to the stiffness support to maintain overall support stiffness. Fig. 2 (a) shows the overall view of the real proposed system.

The proposed system, requires preload control. Support stiffness is positively correlated with preload but also saturates with increased preload [10]. To prevent decreases in overall support stiffness, the preload on the stiffness supports should be maintained within the saturation region. This requires an adjustable preload on the damper support.

To accommodate these requirements, the damper support shown in Fig. 2 (b) was selected as our proposed system. Figure 3 shows damping effects in a one degree of freedom vibration system. For traditional systems, overall support stiffness is decreased by the damper. Subsequently, vibration magnitude is increased for a certain frequency range. Comparatively, the proposed system allows for damping without a concomitant decrease in overall support stiffness. In the proposed system, a Polyisobutylene-base thermoplastic elastomer is used as the damping material. This elastomer—commonly used to prevent furniture from overturning during an earthquake—is most effective for shear stress [11]. Thus, this proposed damper is more effective for the horizontal displacement.

Table 1. Major specifications of the machine tools.

	Small machine	Medium machine	Large machine
Machine type	Horizontal milling machine	NC vertical milling machine	Machining center prototype
Width	0.5 m	2.5 m	1.2 m
Length	0.9 m	1.8 m	2.4 m
Height	1.5 m	2.0 m	2.6 m
Weight	1700 kg	3000 kg	5400 kg
Number of stiffness support	3	6	6

Table 2. Damper conditions in the experiment.

	Small machine	Medium machine	Large machine
Applied damper condition	40×50 mm^2 2 pcs	70×70 mm^2 2 pcs	70×70 mm^2 4 pcs
	40×100 mm^2 2 pcs	70×70 mm^2 4 pcs	70×70 mm^2 6 pcs

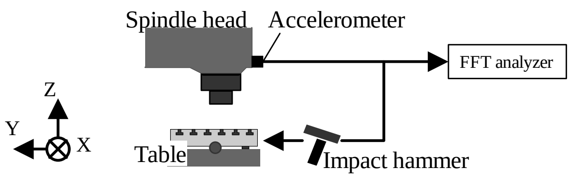

Fig. 6. Experimental setup in the large machine.

3. Experimental methods and results

3.1. Machine tool used in the experiment

In this section, experiments are conducted to investigate relationships between the damper area and the damping in various machines. The proposed damper system is applied to three different machine tools. Machine weight is assumed to affect damping ratio and associated damping coefficient. Figure 4 shows the three machine tools used in the experiment. Major specifications are listed in Table 1. Figure 5 depicts support placement and placement configuration relative to the machine tool footprints.

3.2. Experimental methods

Damper effects are investigated using the impact testing. Frequency response from the excitation force to the displacement is obtained for each machine. An excitation force is applied to each machine with an impulse hammer. Displacement is obtained as the integral value of acceleration obtained by accelerometers at a specific measurement point. Each machine is excited at the table in the Y direction. Acceleration is measured at the spindle. Figure 6 shows the experimental setup for the large machine. Similar setups are used for the other machines.

To investigate the effects of contact area and the machine size, different sizes and quantities of dampers are applied to each machine. Table 2 details the various damper scenarios tested for each machine. Damping effects were expected to increase linearly with increased damper size (contact area).

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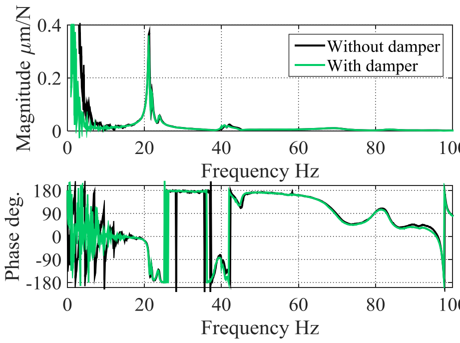

Fig. 7. Frequency response in the large machine with and without damper.

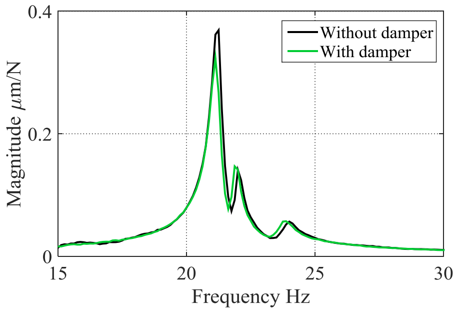

Fig. 8. Magnified large machine frequency response for 15-30 Hz.

Table 3. Obtained modal characteristics of largest resonant peak.

Machine and damper condition	Magnitude in resonate peak μm/N	Largest resonant peak frequency Hz	Damping ratio ζ
Small without damper	1.30	19.6	0.0131
Small 40 × 50 mm^2 × 2	0.70	19.7	0.0270
Small 40 × 100 mm^2 × 2	0.42	19.2	0.0378
Medium without damper	0.049	38.4	0.0587
Medium 70 × 70 mm^2 × 2	0.045	38.3	0.0616
Medium 70 × 70 mm^2 × 4	0.040	37.8	0.0711
Large without damper	0.37	21.1	0.0103
Large 70 × 70 mm^2 × 4	0.33	21.1	0.0108
Large 70 × 70 mm^2 × 6	0.32	21.1	0.0110

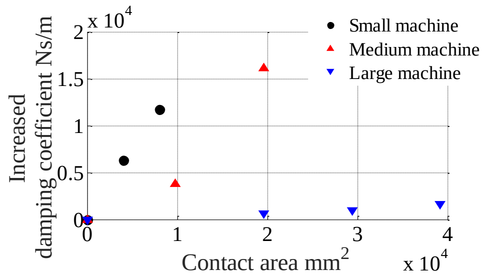

Fig. 9. Relationship between increased damping coefficient and contact area.

According to our previous research, the proposed damper system is more effective with larger preload [8]. Thus, in this research, preload on the dampers are maximized within the saturated region of stiffness support in each machine to maximize the damper effect without decreasing the stiffness. In all conditions, preload pressure is approximately 20–30 kN/m^2.

Sensitivities of impulse hammer and the accelerometer were 0.21 mV/N and 50 mV/m/s^2, respectively. The frequency measurement range was set to 250 Hz in medium machine, and 100 Hz in the large and small machines. The number of sample points was 2048. The results of five repeated measurements were averaged.

3.3. Experimental results

Figure. 7 compares the frequency response of "without damper" and "with damper" conditions in the large machine. In Fig. 7, "70×70 mm^2 6 pcs" conditions is chosen for the "with damper" condition. Figure 8 magnifies the 15–30 Hz frequency range from Fig. 7. According to table 3 natural frequency is unchanged. It means support stiffness is not decreased by the proposed damper. The magnitude of vibration is decreased from 0.37 μm/N to 0.32 μm/N around the first resonant peak, which represents a rocking vibration. These results suggest that the proposed damper system is effective even for the largest machine in the experiment. The proposed damper system is also proved effective for the other test machines.

Table 3 summarizes the modal characteristics of the largest resonant peaks in each condition considered in this experiment. These values are obtained from the curve fit (adapted by the differential iteration method) of the frequency response data. To obtain the damping ratio, the modal mass is assumed as the total mass of each machine because, translation of the entire machine is dominant in the rocking vibrations. According to table 3, natural frequency is largely unchanged by the damper in all conditions. Thus, preload on each damper is well adjusted and support stiffness is maintained in all cases.

3.4. Calculation of increased damping

Damping ratio is affected by the size of a machine. If the same amount of damper is added, the increased damping coefficient should be the same between different machines. In this study, the increased damping coefficient is calculated for each of the condition. To calculate the damping coefficient, the modal mass is again assumed as the total mass of each machine. The modal stiffness is calculated from the modal mass and resonant frequencies listed previously in Table 3. The modal mass, modal stiffness, and damping ratio were collectively used to calculate the damping coefficient.

Figure 9 shows the relationship between the calculated increased damping coefficient and the contact area for each machine. According to Fig. 9, the results confirm a linear relationship between increased damping coefficient and damper contact area; the damping coefficient increases as damper area increases. However, the rate of change differed significantly for each of the machines; the damping coefficient can be different between applied machines even with the same damper contact area and preload. This suggests the influence of a not yet identified confounding factor that should be considered when determining the damping coefficient.

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4. Conclusion

In this study, proposed damper system introduced in our previous study is applied to various machine tools. Results indicate that the damper is effective for machines larger than the machine that is already applied. A linear relationship was observed between the damping coefficient and damper contact area for each machine. However, the rate of change differed significantly for each of the machine; the damping coefficient can be different between applied machines even with the same damper contact area and preload. This suggests a possible confounding factor that should be considered when determining the damping coefficient.

Acknowledgment

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